Dynamic characteristics of helical gear based on Arbitrary Lagrangian Eulerian formulation

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EXTENDED ABSTRACT

1 Introduction

As one of the most basic transmission systems in the industrial field, the gear system has a wide range of applications. However, spur gear systems often induce strong nonlinear vibrations. The transmission of the helical gear system is more stable and it has an irreplaceable position in many engineering fields, but the helix angle will cause the helical gear to generate an axial force during the meshing process, which in turn will generate a tilting moment, imposing higher requirements on the rigidity of the gear shaft. Therefore, it is important to research on the dynamic performance of the helical gear system. Lin et al. [1] applied finite element method to study helical gear system under the static load. In order to accurately simulate the stress distribution of the tooth surface, they conducted highly refined mesh for the gear teeth in meshing area. However, each pair of teeth could come into contact during the meshing process, thus requiring elaborate mesh for whole gears, which greatly increases the degree of freedom (DOF) of system. Tamarozzi et al. [2] applied the static modal switching method to the dynamics simulation of gear system. The modal matrix changes with the contact area, so that it only contains the constraint modes of boundary nodes. The flaw under this method is that the dynamic change of the modal matrix will cause additional vibration to the system. Liu et al. [3] combined Arbitrary Lagrangian Eulerian (ALE) method with Finite Element method in the dynamic simulation of spur gears. Only four engaging tooth-faces were selected as boundary nodes. The modal matrix, stress modal matrix, stiffness matrix, velocity vector and displacement vector are updated during single-tooth contact period to avoid additional vibration[3].

2 Problems and Methods

In order to accurately analyze the dynamic characteristics of helical gear system, it is necessary to establish a highly refined mesh for gears. The large number of DOF in this model will greatly increase the computational costs. This paper aims at analyzing the dynamic behavior of helical gear efficiently and quickly.

Firstly, The ALE method is applied to helical gear system. In ALE formulation, material nodes and mesh nodes are independent to each other, which provides the flexibility of arranging mesh nodes. For most helical gear, the contact ratio $\varepsilon \in (2, 3)$, i.e., only two or three pairs of teeth are in contact simultaneously. Therefore, only three engaging tooth-faces are defined as boundary nodes, resulting in a great reduction in DOF of the system. During the period of double-tooth meshing, the boundary nodes are updated to the on-coming teeth. As shown in Figure 1, the engaging tooth-faces are updated from (3B-3A, 2B-2A, 1B-1A) to (2B-2A, 1B-1A, 30B-30A). At the same time, generalized coordinates, displacement modal matrix, stress modal matrix, stiffness matrix and mass matrix are updated as well. This ALE method can also be applied to gear system with arbitrary contact ratio.

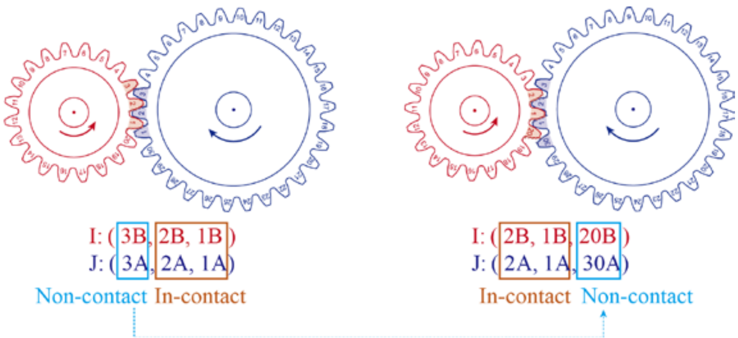

Figure 1: Tooth-change process during double-tooth meshing period

Secondly, low-frequency approximation is adopted to improve the efficiency of helical gear system. It divides the modes of the system into constrained mode $\mathbf{\Phi}_C$ and normal mode $\mathbf{\Phi}_N$. The modal coordinates of the system are $\mathbf{q}_C$ and $\mathbf{q}_N$ respectively, with boundary nodes DOF $\mathbf{u}_B$ and internal node DOF $\mathbf{u}_I$. The frequency of the meshing force is much lower than the natural frequency of gears, so the normal mode $\mathbf{\Phi}_N$ can be ignored. The expression is as equation (1). With this approximation, the dynamic equation can also be simplified by ignoring the inertial force caused by elastic deformation. The dynamic response of helical gear system is quasi-static, so the inertial force engendered by elastic deformation is negligible to the elastic force. That simplifies the expressions of linear term of velocity, square term of velocity and Jacobian matrix.

$$\mathbf{u}=\begin{bmatrix}\mathbf{u}_B\\ \mathbf{u}_I\end{bmatrix}=\begin{bmatrix}\mathbf{\Phi}_C & \mathbf{\Phi}_N\end{bmatrix}\begin{bmatrix}\mathbf{q}_C\\ \mathbf{q}_N\end{bmatrix}=\begin{bmatrix}\mathbf{I} & \mathbf{0}\\ \mathbf{\Phi}_{IC} & \mathbf{\Phi}_{IN}\end{bmatrix}\begin{bmatrix}\mathbf{q}_C\\ \mathbf{q}_N\end{bmatrix}\approx\begin{bmatrix}\mathbf{q}_C\\ \mathbf{\Phi}_{IC}\mathbf{q}_C\end{bmatrix} \tag{1}$$

Finally, the high-efficiency contact detection algorithm is developed to improve computational efficiency. It completes the contact detection through two steps, as shown in the Figure 2. Besides, a simple algorithm for calculating the total length of time-varying contact lines is proposed, with which the dynamic characteristics of helical gears are investigated.

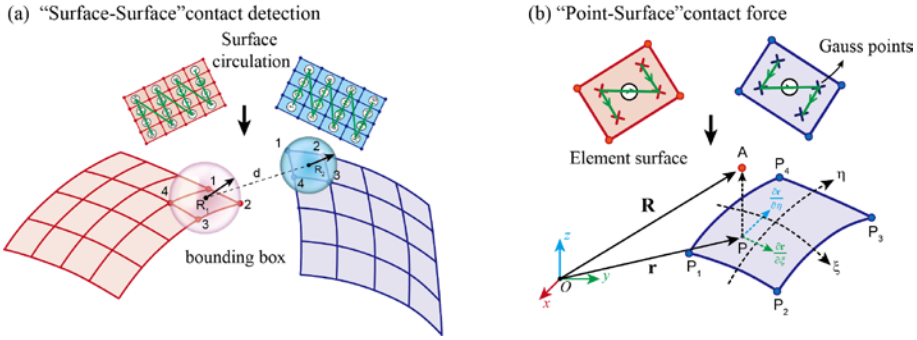

Figure 2: Contact detection algorithm: (a) Surface-surface contact detection, (b) Calculation of point-surface contact force

3 Results

Through a large number of simulation examples with proposed methods, the dynamic characteristics of 4 pairs of helical gear systems with different helix angles are explored. Compared with ABAQUS, the simulation efficiency is improved by about an order of magnitude and the accuracy is highly ensured. Based on the comprehensive analysis of the total length of contact line and the dynamic transmission error, it is found that the transmission characteristics of helical gear depend on the total length of contact line rather that the number of engaging teeth, as shown in Figure 3(a). Besides, the transmission becomes more stable with the increase of the helix angle, but the axial force and the tilting moment will also increase, as shown in Figure 3(b).

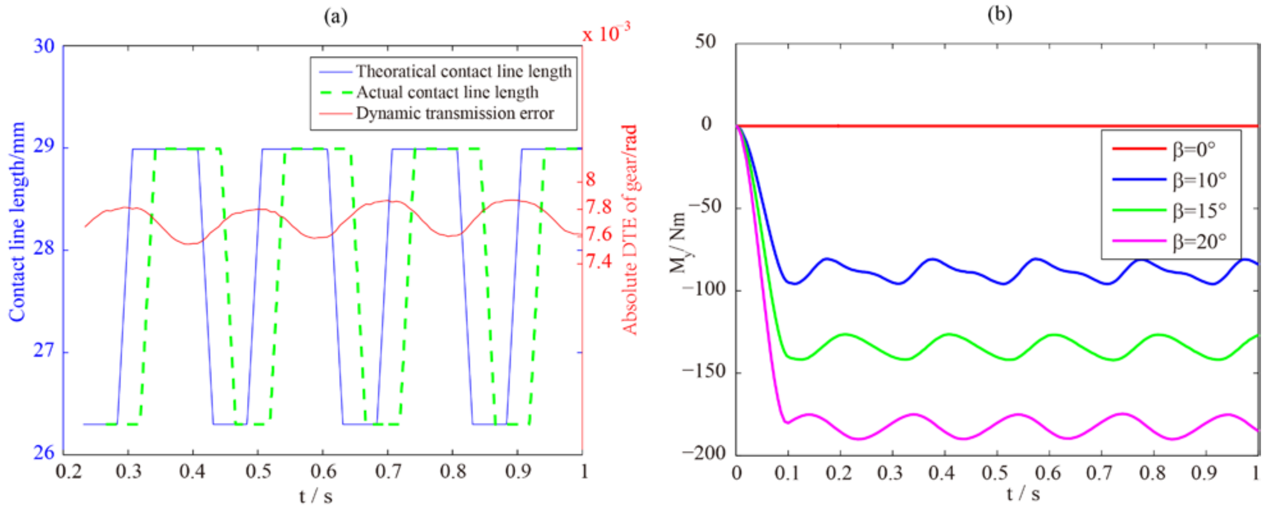

Figure 3: (a) Comparison between dynamic transmission error and total length of contact line when helix angle is 20(deg), (b) Influence of helical angle on moment along Y-axis of gear center

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